PROJECT 8200

ENERGY EFFICIENT COMPRESSED AIR SYSTEMS

SYNOPSIS:

Approximately 10% of all electrical power used in industry is employed in compressing air. Energy efficient compressed air systems play an important role in the effective management of energy.

This report covers the fundamentals of compressed air systems, types of compressors, controls, applications, selection criteria and energy requirements of such systems. Methods of minimizing energy losses and the significance of heat recovery are also discussed. .

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PROJECT SUMMARY

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OBJECTIVES AND NOTES:

Review the fundamentals of compressed air systems; types of compressors, control, applications and installations. Describe the relative characteristics, selection criteria, economics of selection and operation and energy requirements of such systems. Examine the potential for waste heat recovery.

Discuss air leaks. Show one or two examples illustrating proper selection and application of a system.

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1.0 INTRODUCTION

Compressed air is a form of energy that has been used in industries for a variety of purposes. The main reasons for this are that compressed air is intrinsically safe, its application is flexible, and it can be stored without much loss. Its main applications are for operating pneumatic tools and control systems.

Considerable amounts of energy are required to compress air. Approximately 10% of all electrical power used in industry is employed in compressing air. A normal compressed air pressure of 689 kPa (100 psig) requires at least 1 kW of installed motive power to provide a free air flow of 3 L/S (6 cfm). In general, many compressed air systems are inefficient and wasteful, and compressed air is costly. It is for this reason that energy efficient compressed air systems would provide a worthwhile saving in overall energy consumption.

2.0 TYPES OF COMPRESSORS

There are basically two types of compressors (as shown in Figure 1).

2.1 Positive - Displacement (Intermittent Flow) Compressors

In these units, successive volumes of air are admitted to a closed space and elevated to a higher pressure. They are in the following forms:

2.1.1 Reciprocating

The compressing and displacement element is a piston having a reciprocating motion within a cylinder.

2.1.2 Rotary

The compression and displacement is tied-in with the positive action of rotating elements. The following are the different types of rotary compressors:

(a) Sliding Vane

The axial vanes in these machines slide radially in an eccentrically mounted rotor. Gas trapped between vanes is compressed and displaced.

(b) Liquid Piston

Water or other liquids are used in place of a piston to compress and displace the air or gas handled.

(c) Two Impeller Straight Lobe

In these machines, two straight mating lobed impellers trap air or gas and transport it from intake to discharge.

(d) Helical or Spiral Lobe

Air or gas is compressed or displaced by two intermeshing rotors, each with a helical form.

2.2 Continuous-Flow Compressors

2.2.1 Dynamic

Action of the rapidly spinning impeller or rotating vanes impart velocity and pressure to the incoming air.

(a) Centrifugal

One or more rotating impellers, usually shrouded on the sides, accelerate the gas. The flow is in a radial direction. For ranges higher than 1,034 kPa (150 psig) the units are multi-staged.

(b) Axial-Flow

By the action of the bladed rotor shrouded on the blade ends, air or gas is accelerated in an axial direction. In general, these units are suitable for delivering large volumes of air at relatively low pressures. By using multi-stage design, delivery pressures as high as 517 kPa (75 psig) can be obtained.

(c) Mixed-Flow

Impeller form on these machines have combined characteristics of both the centrifugal and axial compressors above.

2.2.2 Ejectors

These machines mix a high velocity gas or steam jet with the inflowing air or gas and convert the velocity to pressure through a diffuser.

Compressors can also be single-stage, two-stage or multi-stage.

Types of Compressors		Methods of Compression
Positive - Displacement (Intermittent -Flow	Reciprocating	Air is compressed as the volume of the enclosed space is reduced.
	Rotary Sliding-Vane	
	Rotary Liquid-Piston	
	Rotary Straight-Lobe	As a definite quantity of air is trapped, pressure is increased and is transferred from the suction intake to the discharge port without reducing its volume.
	Rotary Helical-Lobe	
Continuous-Flow	Dynamic Centrifugal	Velocity and pressure is imparted to the admitted air, through the action of a rapidly spinning impeller or rotating vanes.
	Dynamic Axial-Flow	
	Dynamic Mixed-Flow	
	Ejector	A high velocity gas or steam jet is used to entrain the inflow gas, the velocity of the mixture is then converted to pressure in a diffuser.

FIGURE 1
Types of Compressors

3.0 COMPRESSORS DRIVERS

3.1 Selection Criteria

Drivers are usually selected for the following factors:

- low cost;
- proper coordination with the compressor;
- availability and economy of fuel;
- heat balance;
- power recovery.

For best coordination, it is desirable to select the whole compressed air system including compressor, driver, lubrication system, seal system, coupling and control from the same supplier.

The different types of drivers and their characteristics are as shown in Figure 2 below.

Drivers	Dynamic	Reciprocating	Vane Rotary	Helical-Lobe Rotary	kW Range	Available Speed (rpm) (60 Hz Power)
Induction Motor	C-G	B-C-F-G	B-C	B-C-G	1 to 3,730 (or larger)	3,600/N less 2% *N = 1 thru 8
Synchronous Motor	C-G	C-E-F-G	Rarely	C-G	74.6 to 14,920 (or larger)	3,600/N *N = 2 thru 20
Steam Engine	---	I	---	---	41 to 2,984	400 to 140
Steam Turbine	C-G	G	G	C-G	To 14,920 (or larger)	34,000 to 1,800
Combustion Gas Turbine	C-G	---	---	G	2,238 kW at 10,000 rpm to 14,920 kW at 3,000 rpm (305 m altitude and 27°C)	
Integral Gas Engine	---	I	---	---	From 63 kW at 600 rpm between 3,730 kW and 4,476 kW at approx 300 rpm	
Integral Diesel Engine	---	I	---	---	From 74.6 kW at 600 rpm to approx 1,300 kW at lower rpm	
Coupled Gas or Diesel Engine	---	C	---	---	From 74.6 kW at 600 rpm to 3,730 kW or more at approx 360 rpm	

Key: B - Belted
C - Coupled
E - Engine-Type Direct Connected
*N - No of pairs of poles
F - Flange-Mounted
G - Geared
I - Integral

FIGURE 2: Common Compressor Drivers and Their Characteristics

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3.2 Two Common Types of Compressor Drivers

3.2.1 Squirrel-cage induction motors are widely used, especially in the 0.75 to 150 kW (1 to 200 hp) range. They become less economical when they exceed 746 kW (1,000 hp). These motors are simple, rugged, reliable, reasonably efficient and have no rotating windings, slip rings or commutators, generally associated with other motors. The disadvantage of the squirrel-cage induction motors are their lagging power factor. Most centrifugal compressors are driven by 2-pole, 60 Hz, electric motors for speeds up to 3,550 rpm. Increasing gears are used for higher speeds. For slower speeds, 4, 6 or 8-pole motors are used.

3.2.2 Synchronous motors are generally used to drive large reciprocating compressors because of their multiple selection of slow speeds which permit direct connection. They are the least costly electrically operated prime drivers when sizes above 150 kW (200 hp) are involved. They are less reliable than squirrel-cage motors due to their use of a wound rotor to which direct current excitation must be applied, usually through rotating slip rings. Synchronous motors can operate in a leading power factor mode. It is common in industry to use synchronous motors to provide power factor correction.

4.0 CHOICE OF COMPRESSORS

Air compressors generally operate at pressures up to 3,447 kPa (500 psig), with the majority in the range up to 862 kPa (125 psig). When choosing a compressed air system, factors for optimum energy efficiency should be considered.

4.1 Power Consumption

The reciprocating compressor is the most efficient, followed in order by the centrifugal, the vane, and the helical-lobe rotary units.

4.2 Capacity Control

The reciprocating compressor is best suited to handle a variable air demand from zero to full capacity.

4.3 Load Factor

This is the ratio of actual compressed air output while in operation to the rated full load output. The desirable load factor lies within the range of 50% to 80%. This mode of operation results in:

- more uniform pressure, even during peak demand periods;

- a cooling-off period. Equal compression and shutdown times are usually required for air-cooled units operating at pressures higher than 1,379 kPa (200 psig);
- less maintenance caused by early deposits on valves and other parts compared with substained full load operation;
- extra capacity for increased load.

4.4 Multi-Staging

When pressure required is 689 kPa (100 psig) or higher, air-cooled compressors with two or more stages, would run cooler due to multi-stage intercooling. They are of higher efficiency and consume less power than their single-stage counterparts. Minimum energy is used during no-load condition. A cooler running compressor would reduce deposits on valves and other parts and require less maintenance. However, if atmospheric conditions and air temperature permits, heavy duty water-cooled units would run more economically than the air-cooled units. Water-cooled units normally run at considerably lower speeds and temperatures.

5.0 THE DISTRIBUTION SYSTEM

In a compressed air distribution system, there are two ways of locating compressors. One is the centralized arrangement, with all compressors at one location. The other is the decentralized system with compressors at the various centres throughout the plant. The various centres may or may not be interconnected.

5.1 Centralized Distribution

The principal advantages of the central plant include the possibility of using larger, more efficient units; reduced costs for bringing in the various utilities; better maintenance and supervision at lower cost; and better balance between air supply and demand. The central plant may be at a greater disadvantage in the maintenance of proper air pressure, although this should be no problem with a properly designed distribution system.

5.2 De-Centralized Distribution

The principal advantage of the de-centralized system is its flexibility. It is simple to add units where needed and piping runs are short.

Separate smaller units are frequently used to supply necessary or emergency services during weekends and at night, thus permitting shutdown of the larger units during periods when their load factor and efficiency would be materially lowered.

A well laid out and maintained distribution system should deliver 689 kPa (100 psig) compressed air to the point of application with a limited pressure loss, preferably at a pressure close to 620 kPa (90 psig).

Most distribution systems will consist of a main line from which branch mains can supply different buildings, departments or sections.

Other reasons for dividing the duty between several compressors is that the compressors can be alternated as standbys for periodic maintenance. In some cases where loads are widely scattered in the plant, it is best to install a series of two or three compressor stations at strategic points, coupled together by a ring main. This would reduce line loss.

5.3 Pipe Sizing

Figure 3 serves as a guide for selecting pipe sizes to minimize pressure drop under the design conditions. It is advantageous to use pipes of one or two sizes larger than required for better pressure maintenance and for future expansion.

5.3.1 Main Line

The main line should be sized for the average rate of all air uses including losses through fittings. Pressure drop in this section should be no more than 21 kPa (3 psig). For pressure drop of less than 10% of the initial pressure (P_1) use Figure 3, otherwise use the following formula to calculate correct line pressure drop:

$$\Delta_{PL} = P_1 - \sqrt{P_1^2 - 2KL}$$

where $$K = \frac{\Delta_{p100}}{S}\left(P_1 - \frac{\Delta_{p100}}{2}\right)$$

Δ_{PL} = required total pressure drop in kPa(psig)
P_1 = initial pressure in kPa (or psig)
L = pipe length in meters (or feet)
S = 30.48 m (100 ft)
Δ_{p100} = pressure drop per 30.48 m (100 ft) from Figure 3

5.3.2 Branch Mains

Demand variation is less in branch mains and should therefore be sized for their maximum flow. This may run 150 to 175 percent of the average, depending upon the number of outlets to be in use at any one time. Pressure drop in this section should be no more than 21 kPa (3 psig). Use Figure 3 to determine pressure drop.

Air Flow		Nominal Pipe Size*									
CFM	(L/S)	1/2" (15 mm)	3/4" (20 mm)	1" (25 mm)	1½" (32 mm)	2" (50 mm)	2½" (65 mm)	3" (80 mm)	4" (100 mm)	5" (125 mm)	6" (150 mm)
10	(4.72)	2.62	-	-							
20	(9.44)	9.72	2.34	0.69							
30	(14.16)	21.03	-	-							
40	(18.88)	36.61	-	-							
50	(23.60)	56.54	13.31	3.79							
100	(47.20)		51.92	14.41	1.65						
150	(70.80)			31.72	3.65	1.03					
200	(94.40)			55.50	6.34	1.79	0.76				
300	(141.60)				14.13	3.93	1.59				
400	(188.80)				24.75	6.76	2.83				
500	(236.00)				38.54	10.41	4.34	1.38			
1000	(472.00)					40.68	16.75	5.24	1.31		
1500	(708.00)						36.96	11.58	1.38	0.90	
2000	(944.00)						64.81	20.62	5.17	1.65	
3000	(1416.00)							46.26	11.31	3.52	1.38
4000	(1888.00)							82.05	19.99	6.21	2.41
5000	(2360.00)								31.03	9.65	3.79
6000	(2832.00)								20.20	13.79	5.38
7000	(3304.00)								60.47	-	-
8000	(3776.00)								79.29	24.34	9.51
10000	(4720.00)									37.71	14.69

* Size (mm) in brackets are the "diametre nominal" equivalent to the imperial system.

Note: To estimate pressure drop multiply above values by number of 30.48 m (100 ft) lengths plus the equivalent 30.5 m lengths for fittings. For pressure drops above 10% of initial value, use formula in Section 5.3.1.

FIGURE 3: Pressure Drops in kPa For Each 30.5 m (100 ft) of Straight Pipe
At 689 kPa (100 psig) Line Pressure

5.3.3 Feed Lines

Feed lines from the branch main to the floor should be sized for the maximum flow. These lines are usually short and pressure drop should be less than 7 kPa (1 psig).

Figure 4 illustrates various piping arrangements. Section A shows a branch main with air feeding in one end and outlets at intervals. This type is frequently overloaded by over-extention. This condition can be relieved by either extending the main to the midpoint of the branch or, by running a parallel branch and taking half the outlets off the new line, and tieing both lines together at the far end.

Better distribution can be obtained if the ends of two lines are joined to form a loop (see Section B), thus giving two-way air flow to feed a point of heavy demand.

5.3.4 Air Receiver

For heavy intermittent demands, possibly at the end of a long small line (see Figure 4 Section C), serious pressure drops can be reduced by installing a simple air receiver as near as possible to the demand point, thus storing air where it is needed. For example, say 94.4 L/S (200 cfm) free air is required for two minutes every ten minutes. It would overload the feeder line and cause excessive pressure drop to draw at a rate higher than 35.4 L/S (75 cfm). Therefore, 59 L/S (125 cfm) or a total for two minutes of 118 L/S (250 cfm) free air, must be stored. Line pressure (that which is in the receiver when the equipment is not running) is 662 kPa (96 psig) and allowable maximum pressure drop in the receiver when equipment operates is 69 kPa (10 psig). The 118 L/S (250 cfm) free air must be taken from the receiver storage within this pressure drop. The receiver volume is:

$$V = \frac{59 \times 2 \times 60}{69} \times \text{atmospheric pressure in kPa}$$

$$= \frac{7080 \text{ L} \times 101.325 \text{ kPa}}{69}$$

$$= \underline{10{,}396.83 \text{ litres}}.$$

This receiver would be about 1.5 m (60 inches) diameter by 6.0 m (19.7 feet) long. The receiver pressure would be restored during the period the equipment was idle without appreciable effect on the main line. Without the receiver, the pressure at the equipment would fall too low and its operation would be impaired.

Fig. 4 Various types of compressed air distribution systems.

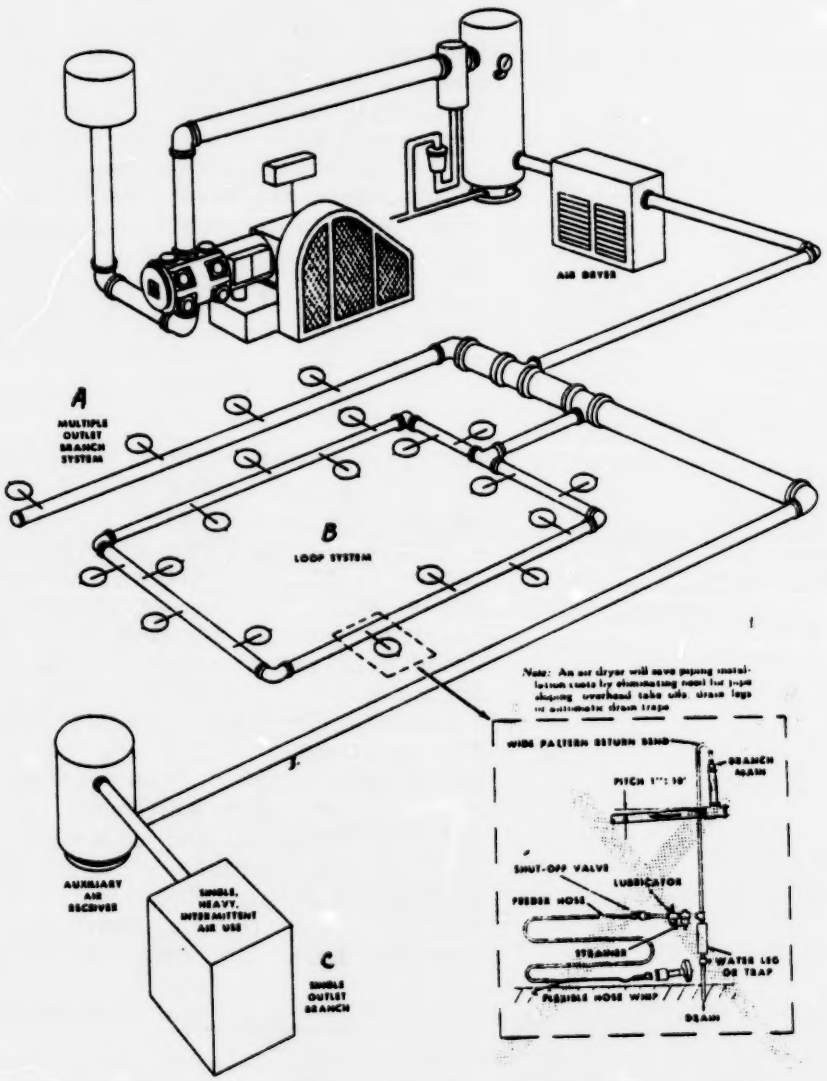

Fig. 5 The proper method to connect a pneumatic tool to a distribution system.

6.0 ELEMENTS OF CONTROL SYSTEMS

The following diagram illustrates the fundamental operations of a compressor control system.

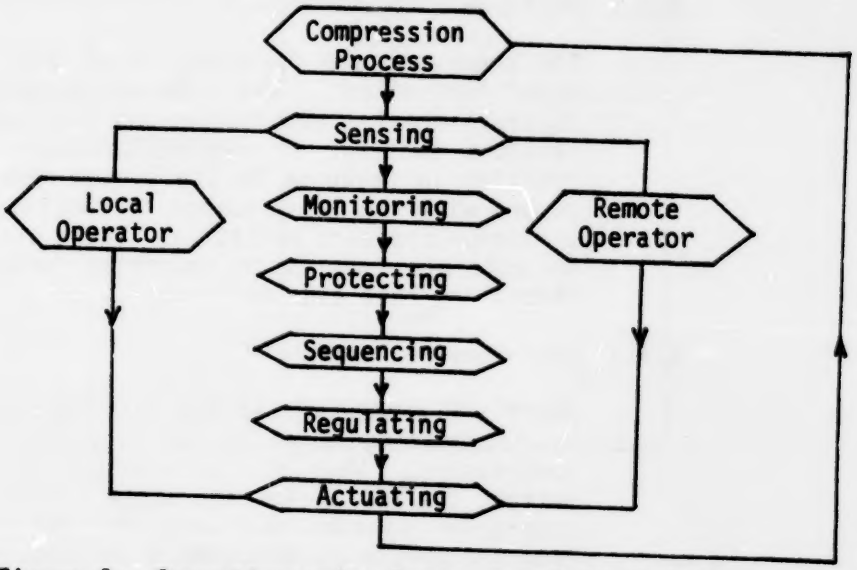

Figure 6: Operations of a Compressor Control System

6.1 The sensors gauge any critical variations and send signals to the controls by electric, hydraulic or pneumatic means.

6.2 The monitors, having received signals from the sensors, will announce any fault by alarms and/or warning lights. In some cases, they will also activate protective or shutdown devices.

Monitoring and protective panels are normally used for motor driven compressors.

6.3 Compressor controls can be classified into four basic types:

Type A - the one mode of operation is ON/OFF. With this type of control method, output is either 100 percent or zero. It is accomplished by full load or unloaded operation or by start and stop operation.

Type B - compressors with two or more discrete part-load steps.

Type C - compressors with continous throttling.

Type D - unloads only partially. Output is then reduced to zero as in Type A.

6.4 Sequence controls can apply to machines in one or more locations. The following paragraphs describe several compressor sequence controls.

6.4.1 Multiple Compressor

The sequences are "pressure only" and "pressure plus time" controlled". The simplest sequence is pressure controlled only; compressors are started and stopped by pressure switches at various pressure levels. They function in response to the system demand and meet the demand with the least number of compressors on-line, and a minimum pressure variation. There is a basic trade-off in multiple compressor sequences between a narrow throttling band and low frequency of stops and starts.

6.4.2 Anti-Recycle Timers (ART)

These are commonly used for limiting starting frequency by introducing a delay in stopping or starting compressors. When rising pressure trips a pressure switch, the ART begins its delay function and allows the compressor to stop only when the ART times out. This limits startups by delaying stops, but unfortunately it can also eliminate stops altogether and may keep a compressor running indefinitely. A momentary system pressure drop is enough to reset the pressure switch and the timer before the ART can time out.

An alternative control system begins timing whenever the compressor starts and allows it to stop any time after the ART times out. This method works independently of any pressure signal.

One problem with the commonly used ART method is that a compressor may continue to run on ART at high pressure. This situation may occur with type A or D controls, (see Section 6.3), which either cannot unload or only partially unload. These types of controls allow pressure in the system to increase to the point at which the compressor shuts down on motor overload, or a sytem pressure relief valve lowers the pressure.

6.4.3 Pressure-Only Control

These require consideration of the plant electrical system inrush capability. It can be seen in Figure 7 that if a number of compressors are sequenced at narrow pressure intervals, a sudden drop in pressure could require several compressors to start within a very short time and cause a substantial voltage drop.

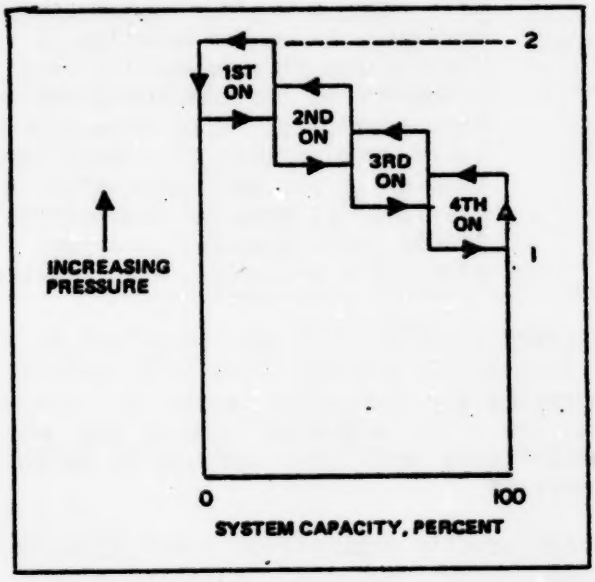

Figure 7: Pressure versus Capacity for a Typical Four-Compressor Sequential Operation With On/Off Control

6.4.4 Sequencing Timer Settings and Logic

By using variable timers, optimum settings can be made for various load conditions. Time settings can be arranged proportional to the pressure.

Another consideration is the starting sequence. For systems in which all compressors are of equal size, the usual concern is with equalizing running time for maintenance purposes. For systems that use unequal size compressors, the sequencing depends on the plant load. The compressors should be started in a sequence that allows the minimum power to satisfy the demand.

For pressure-only sequences, each pressure switch can be wired into a selector switch or similar device that transfers the starting of various compressors to each switching point. They are called base load transfer switches. Because the possible combinations add up very quickly when there are more than three or four compressors, a limited number of sequences, usually limited to the number of compressors, is used.

With more than four compressors, the wiring becomes complex. An easier solution is to use the high degree of logic processing capability in a programmable controller, transferring the complexity out of the wiring system and into a program. With this type of equipment, all possible combinations of control are possible. The logic complexity becomes inexpensive in these systems, because they are designed for mass-memory computer-type applications. This approach provides for unlimited manually adjustable sequences, when required.

7.0 AIR LEAKS

Compressed air leaks can result in a serious energy loss. Air leaks are often tolerated because they are not always hazardous. Leakage tests have shown wastage of up to 20 percent of the air compressed.

Leakage usually occurs from a multitude of small openings. Each may be minor, but the cumulative effect could be great. An approach to minimizing leaks is outlined below.

- Inspect the stem packing of every valve and cylinder in the system, and repack when necessary;
- repair or replace leaking shutoff valves, couplings and pipe joints;
- repair leaking valves on tools; tools should not be left live on the line when not in use, even though used only occasionally;
- standardize on good hose clamps; leaks are more frequent when make-shifts are used;
- standardize on good blow guns - homemade types often leak. Also reduce air pressure for blow guns to increase safety and to save air;
- install condensate separators and drains at branch ends to eliminate the need for blowing down lines to clear water; and
- use good quality air hose to eliminate breaks.

The cost of a small leak in terms of dollars wasted over a period of time can be considerable as shown in Figure 8. Leakage is assumed to originate from a sharp-edged orifice continuously at 689 kPa (100 psig).

Diameter of Opening mm (in)	Air Leakage L/S (scfm)	Cost of Wasted * Energy Per Year
0.4 (1/64)	0.2 (0.4)	$ 21
1.6 (1/16)	3.1 (6.5)	$ 342
3.2 (1/8)	11.8 (25.0)	$ 1,314
6.4 (1/4)	47.2 (100.0)	$ 5,256
9.5 (3/8)	106.2 (225.0)	$11,812

* Based on 75% operating time - 6,570 hours/year
746 W to produce 2 L/S air flow and energy cost of 4¢/kW.h.

FIGURE 8
Cost of Wasted Energy Per Year For Different Sized Leaks

8.0 HEAT RECOVERY FROM A COMPRESSED AIR SYSTEM

When expanding or installing a new compressed air system, heat recovery should be considered. Almost all the energy used in compression is converted into heat. Theoretically, heat extracted by the cooling of compressed air is equal to the energy input to the compressor if the air temperatures before and after compression are equal. Figure 9 illustrates the energy flow of a compressed air system.

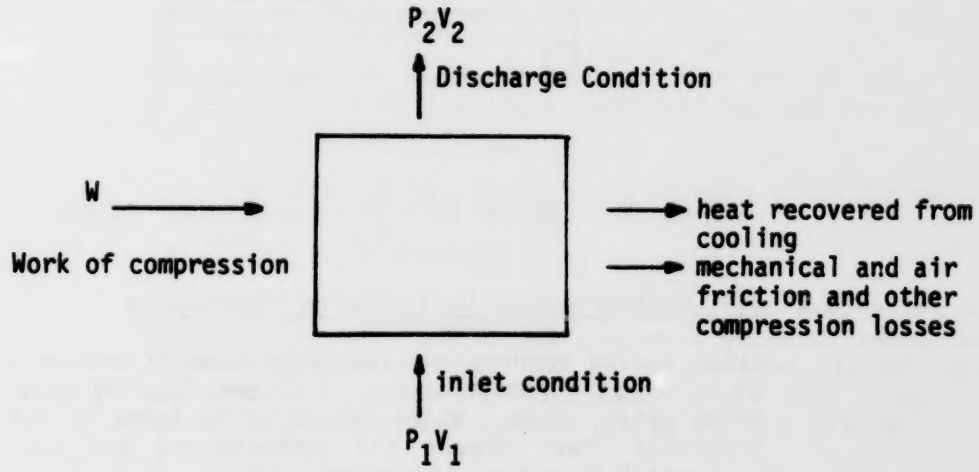

FIGURE 9: Energy Flow of a Compressed Air System

In a standard compressed air system, energy input to the compressor by the electric motor is successively reextracted by compression stage cooling to maintain reasonable working temperatures by the intercooler to improve efficiency and by the aftercooler to remove moisture from the compressed air.

8.1 Heat Recovered from Air-Cooled Compressors

Air cooled compressors should be used if recovered heat is to be used in adjacent areas. Hot air leaving the intercooler and the aftercooler can economically be ducted to adjacent rooms for space heating. This will provide heat recovery of about 85% of the input energy to the compressor. As shown in Figure 10, a recirculation flap valve could be used to maintain constant room temperature for fluctuating compressor loads.

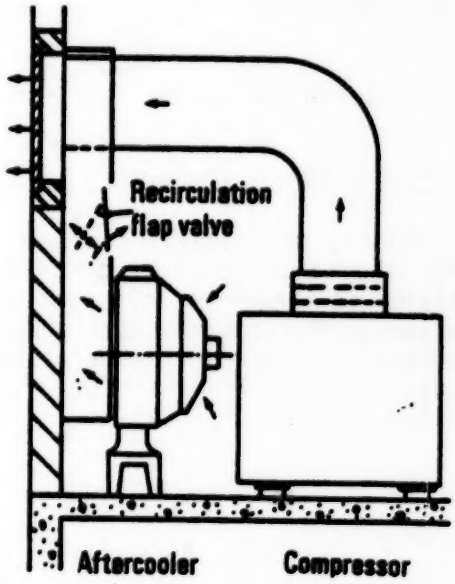

Figure 10: Typical Air-Cooled Compressor System

8.2 Heat Recovered from Water-Cooled Compressors

Water cooled compressors should be used if recovered heat is to be used in remote areas. A closed cooling water circuit saves water costs. Water cooled units normally run at considerably lower speeds and temperatures and are more economical than the air cooled units.

Figure 11 shows a typical installation for a water-cooled compressor system.

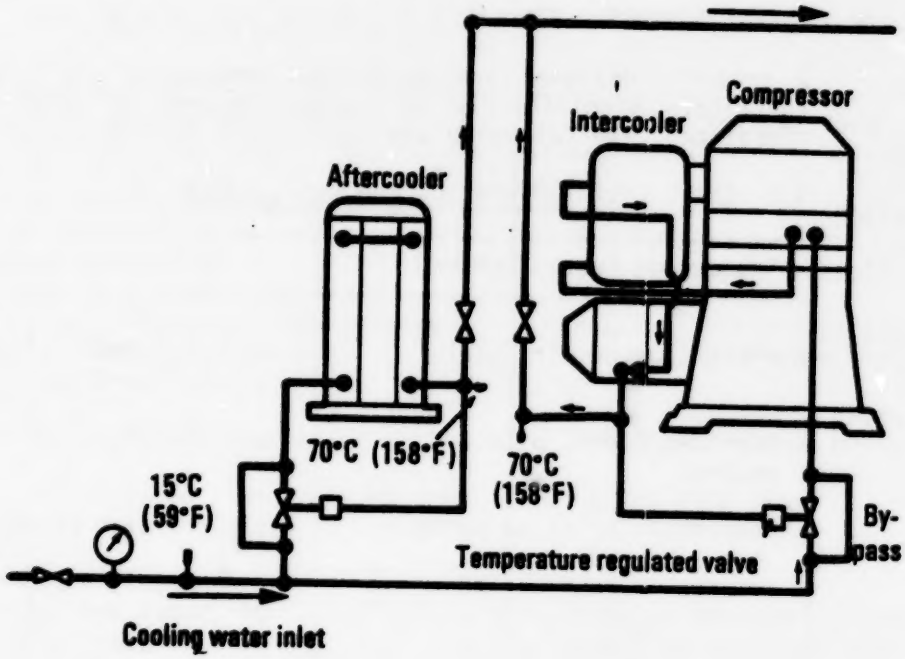

Figure 11: Typical Water-Cooled Compressor System with Water Recirculation Feature

8.3 The Value of Heat Recovery

Consider a compressor with a motor input of 100 kW running at full load for 75% of the time (6,570 hcurs per year), and with 85% heat recovery. At an average electrical energy cost of 4¢/kW.h, this will provide a return of $16,754 per year from heat recovery i.e.:

0.75 x 0.85 x 100 kW x 6,570 hrs x 4¢/kW.h = $16,754/year.

8.4 Tax Incentive Information

It should be noted that under the Canadian Income Tax Act Registration No. "SOR/80-935" dated 5 December 1980, heat recovery systems qualify for an accelerated capital cost allowance for corporate taxation. Under the "Class 34 budget", heat recovery systems can be written off over:

- three years if acquired after 12 November 1981 but before 31 December 1984 (up to 25% in the first year, up to 50% in the second year and the balance in any subsequent year).

For all new installations regulations permit a three year "write-off period". It should also be noted that the change is not retroactive.

9.0 ENERGY SAVINGS CHECK LISTS

A properly designed and maintained compressed air system can contribute significantly to improved energy efficiency of an operation. The following are check lists for corrective actions.

9.1 Check List - Operation and Maintenance

Sources of Energy Losses	Corrective Actions
- dirty plugged air intake	- maintain cool, clean sheltered free-flow inlet
- corroded pipes, pits and pockets	- keep inlet and outlet air dry
- compressor belts slipping	- tighten driving belts
- compressor over-worked	- re-schedule work load to shave off peaks and idlings - on auto start/stop running avoid more than 4 starts/hour
- over pressure	- reduce air pressure to the minimum acceptable level
- scaled or blocked cooling ducts	- regular drainage of intercoolers and aftercoolers - clean coolers on regular basis for optimum heat transfer
- excessive line pressure drop	- replace clogged filters
- air leaks	- inspect the stem packing of every valve and cylinder in the system, and repack when necessary - repair or replace leaky shutoff valves, couplings and pipe joints - repair leaking valves on tools - standardize on good hose clamps - install condensate separators and drains at branch ends to eliminate need for operators blowing lines to clear out water - use good quality air hose to eliminate breaks

9.2 Check List - Retrofit

System Deficiencies	Corrective Actions
- inefficient improvised tools and nozzles	- use effective engineered tools, blow-guns and nozzles to suit specific applications
- air intake located indoors in a warm location	- relocate the air intake to outdoor for cooler air
- air mains partially plugged by condensed moisture	- install automatic drain valves for sloped mains
- excessive line pressure drop	- resize pipe lines - minimize bends - replace improperly sized couplings and fittings
- poor monitoring system	- install pressure gauge on the receiver - install water temperature gauges in the compressor cooling jacket and the after cooler - install air temperature gauges at the outlet of the compressor and in the receiver - install pressure gauges at selected points along the line to monitor effectiveness of devices and to diagnose sectional air leaks - install regulators to achieve minimum acceptable pressure levels

10.0 RECOMMENDATIONS

(a) Use multi-stage units if required pressure is 689 kPa (100 psig) or greater for the energy saving benefits.

(b) Choose compressor with the highest ratio of free air delivery (L/S air flow) to brake horsepower.

(c) Install air receivers for air demand surges.

(d) Specify the guaranteed air delivery required (i.e. quantity of free air flow required) when ordering a compressed air system.

(e) Maintain operating air pressure at the minimum required.

(f) Perform regular maintenance to minimize air leaks.

11.0 REFERENCES

(a) Compressed Air and Gas Handbook - Compressed Air and Gas Institutes, Third Edition, 1961.

(b) Industrial Air Compressors by F. G. White, 1967.

(c) Operation and Maintenance of Air Compressor Plants - Department of Navy Bureau of Yards and Docks, Washington, D. C, 1964.

(d) New Compressed Air and Gas Data, Second Edition, 1971, by Ingersoll-Rand.

(e) "Energy Management for Industrial Plants", Lecture by L. W. Pasternak, Manager, Atlas Copco Canada Ltd., October 1980.